



ENGINEERING DESIGN

Flap Actuator 2

Abstract

Design calculation and analysis for a light Aircraft.

58167

HND/C MECHANICAL ENGINEERING

ENGINEERING DESIGN

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NAME: GERAINT WILLIAMS

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LECTURER: M LEWIS

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Executive summary

Aircraft flap actuators are mechanical systems used in adjusting the flap rates on aircraft wings. Flaps provide the adjustable control surfaces on the rear edges of the aircraft's wing that are retracted or extended to the adjacent surface area and the plane's wings' profiles to necessitate efficient flight at a lower speed. Flap actuators are typically lead screw-type mechanism powered by either hydraulic or electric motor systems. The flap actuators are mounted on a fixed position within the plane's wing. Within the flaps' systems are the movable nuts on the lead screw having an integral bracket attached to the flap assembly. The design of these systems requires well-thought-out procedures and specifications of the various elements that make up the aircraft's flap actuator system. Therefore, this report presents a product design specification for designing a light aircraft flap actuator with detailed specifications on the power screws, gears, and shafts. Power screws are utilized in this design since it provides the best desirable linear and uniform motion and necessitates the transformation of rotary motion into linear motion. Gearbox and Electromotor are utilized in the design configuration to attain an effective torque application from Electromotor to power screw for activation of the flaps.

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Introduction

In the development of aircraft, every component is designed precisely according to predefined specifications. Parts, members, and mechanisms involved are subjected to electronic and mechanical analysis to identify and derive the most desirable electromechanical features for the aircraft's design. The process of deriving the desired feature for the design purpose is referred to as the product design specification. For instance, naval aircraft, being a perfect example of a light spacecraft, the design requires accurate and precise specifications on the design parts for efficient operations (Qiao et al. 2018, p.140). The product design specification is essential in every mechanical product development as it guarantees safety while in operations. The product design specification is integrated within the design stages, and all the desired specifications have to be harmonized to build into one aircraft (Taylor, and Panella, 2017, p.56). Therefore, this report presents the product design specification analysis for designing a flap aircraft actuator for a light airplane such as Naval aircraft.

Flap Actuators

The aircrafts actuation systems' design depends on various factors such as the specified interface, performance requirements, and desired configuration of the aircraft's control unit. Power drive units can either be electromechanical or hydro-mechanical and either on a dual or single motor type (Qiao et al. 2018, p.140). In the electromechanical power drive unit's dual-motor type, the output motor can either be of speed summed or torque summed mode. The flap actuators' shafts consist of torque tubes that connect the power drive unit's output with the left and right-wing actuators. However, the small commercial airplanes' configuration often utilizes flexible drive

shafts reciprocating at considerably higher speed instead of rigid low-speed shafts (Taylor, and Panella, 2017, p.131). Aircraft actuator systems are normally based on ball-screws and are linearly designed; flap actuators, on the other hand, utilizes irreversible ACME screws while other flap actuators are designed in rotary reversible ACME screws.

It is unrealistic to have uniform parts or components design specifications for all types of airplanes. Components and systems to be used in any specific aircraft are determined by various factors such as size, capacity, and purpose of aircraft operations' flight and speed. For instance, it is quite a challenge for aircraft to pose the comfortability of passenger transport and simultaneously have the manoeuvrability of the plane (Qiao et al. 2018, p.140). The class and type of the aircraft determine strong the various components and parts of the aircraft should be built in order to attain the desired safety factor.

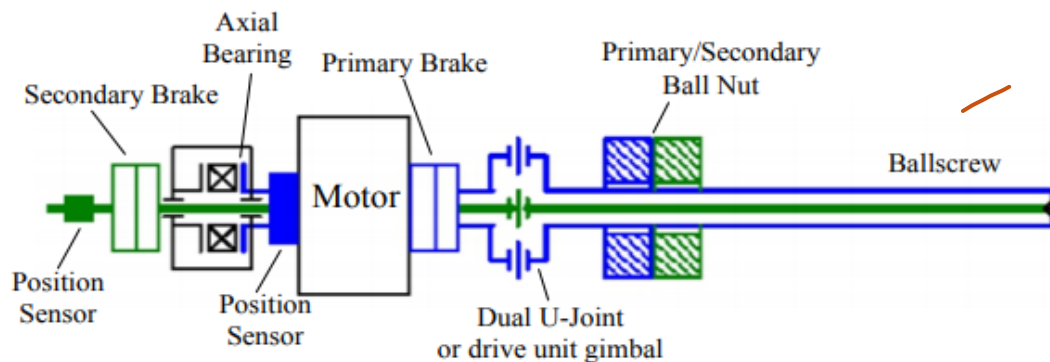


Figure 1: Electro-Mechanical Actuator Flap application

A fixed airframe design consists of wings, fuselage, flight control surface, stabilizers, and landing gears. While the rotary-wing airplanes consist of the rotor assembly, fuselage, tail rotor assembly, and landing gears. As the name suggests, flap actuators' main purpose in aircraft power drive units is to prevent asymmetry between right and left wings flap in the event of shaft

misalignment or failure (Taylor, and Panella, 2017, p.56). The flap actuators are equipped with an asymmetry monitoring system to detect the failure on the plane's wings and automatically activate flap actuator systems to take control of the surface on the command position.

Actuator specification

Electro-Mechanical Actuator (EMA) Configuration

The project considers the design of an electromechanical flap actuation system based on a BLDC motor position servosystem consisting of the drive torque element and a high precision power screw in a closed-loop control position for optimal accuracy of the system. The design incorporates a redundant RVD feedback system for precision and position measurement while in operations. The staple of the actuator system is electromechanical actuation based on the electric motor's rotation with the integration of a power screw within the rotor for the conversion of rotational motion into linear movement of the flap (Giurgiutiu, Chaudhry, and Rogers, 2015, p.43). The primary and secondary double load path architecture is considered to provide double safety with the back brakes systems. The feedback signals are measured by the redundant transducer incorporated in the EMA system.

The specifications relate to light, medium-size aircraft considering the load cycle and capacity of the plane. For instance, load cycle development procedures have been utilized in the analysis of the torque, stress, and strain requirement for various plane components. The component specifications for the proposed aircraft flap actuator are as illustrated in the subsequent sections.

Electro Motor (BLDC Motor)

The design utilizes an electromotor, typical rare earth permanent brushless magnet type. The Electromotor design has been completed in the CAD software program, and the design is based on the surface-mounted radial flux PM Electromotor with specifications derived through electromagnetic motor expressions and equations (Qiao et al. 2018, p.96). The type of motor configuration, motor specifications, material type, and other assumed design data is provided as the input parameters. For instance, the rotor shape and stator dimensions are computed with respect to motor performance rating.

<i>Motor specifications</i>	
No. of Phase	5
No. of Poles	18
<i>Phase Current</i>	
Healthy Operation Mode	4.4
One open-phase fault	6.3
Two open-phase faults	12.4
Speed and torque for healthy operation mode	18.0 Nm at 600 rpm
Speed and torque for one open-phase fault	18.1 Nm at 600 rpm

Speed and torque for two open-phase faults 17.8 Nm at 400 rpm

The motor specification and performance have been computed with consideration to operating load at optimal speed, the efficiency of mechanisms, and frictions involved. The main source of resistive torque for the system is presumed to emanate from; no-back drag torque, screw shaft seal, torque from rotor seals and bearings. The desired motor torque is computed through

$$T_{MOT} = \frac{L_{OP} \cdot P}{2\pi \cdot \eta_{BS}} + T_{DSS} + T_{DNB} + T_{DROTOR}$$

And the desired power input is calculated through rated speed motor efficiency and the obtained motor torque. The formula for power input is as shown.

$$P_{IN} = \frac{n \cdot T_{MOT}}{\eta_{MOT}}$$

Where

P – Power Screw Pitch

T_{MOT} – Motor Torque

L_{op} – Operating Load

η_{MOT} – Motor Efficiency

η_{BS} – Power Screw Direct Efficiency

T_{DNB} - No-back Drag Torque

T_{DSS} – Bearing Drag Torque and
Screw Seal

n- Speed

Actuator optimization

The effects of mechanical constraints observed on the first feasibility test on the torque versus pitch requirement for optimization of the flap actuator depend on the electromotor design. Therefore, the design has been refined through optimization procedures to maximize the torque to current ratio at the base speed. But because the outer stator diameter and stack length are constrained, the variables are reduced; therefore, the major focus was considered on the stator winding and stator dimensions. Design optimization challenges were solved through the Multi-Objective Genetic Algorithm (MOGA) (Qiao et al. 2018, p.75); the algorithm has proved to be the best alternative choice for effectively solving design optimization challenges emanating from real-world applications. It has an elitism feature that enables optimization with no involvement of premature convergence to optimal local frontiers.

Electro-Motor Selection

The process of motor sections in electromechanical actuators is quite similar to one for pneumatic or hydraulic actuators, with only a slight difference in specifications. For instance, in the electromechanical flap actuator's motor selection, the motion profile is a key consideration since it establishes the required time and velocity as well as torque and the desired travel distance for the flap shafts. It also helps in determining the required maximum stroke as well as the speed range for the shaft. The load calculation is another important parameter considered in the selection of motor for the system. Load estimations include externally applied loads, friction loads, and gravitational load (Bellini, Colli, and Dragoni, 2019, p.67). However, the calculation of these load

components depends on the actuator's orientation and configuration, whether it is vertical or horizontal.

Besides load calculations, the resting time ratio to operating time, referred to as duty cycle, is equivalently essential in motor selection. The duty cycle is often expressed in terms of percentage. It's important in the design process as it helps the technicians estimate the flap system's life requirement, prediction of premature failure of parts, wear rate and overheating on the bearing and joints. Knowing the actuator's precision and positional accuracy are key considerations in the design since these two parameters dictate the safety requirement for the design (Qiao et al. 2018, p.54). For the selection of electromechanical motor for this particular project, all of the above-mentioned aspects were scrutinized, and calculations were computed to come up with the most desirable specifications for the motor for the accomplishment of the design objective.

Initial assumptions

The following assumptions were considered during product design specifications for the proposed aircraft flap actuator.

- i. The surface control system experiences non-degradation during gust, thus prevents mechanical damages to the system.
- ii. The actuator has a fail-freeze; it implies no dissymmetric deflection during power loss from the power screw system.
- iii. There are no jamming issues with the surface control systems during actuator motion.

Research and design development

The methodology adopted in the proposed linear electromechanical actuator design is well researched to identify the major challenges with the currently existing flap actuators. The research findings have been used to develop a more feasible and optimized EMA flap actuator for light aircraft (Taylor, and Panella, 2017, p.56). The developed actuator is for business jet; thus, it is designed to actuate the airbrake systems as well. This implies that primary flight control mechanisms of commercial airplanes are not considered in the design development. Instead, the surface control is to be symmetrically controlled with the displacement curve.

Developed designs in CAD software

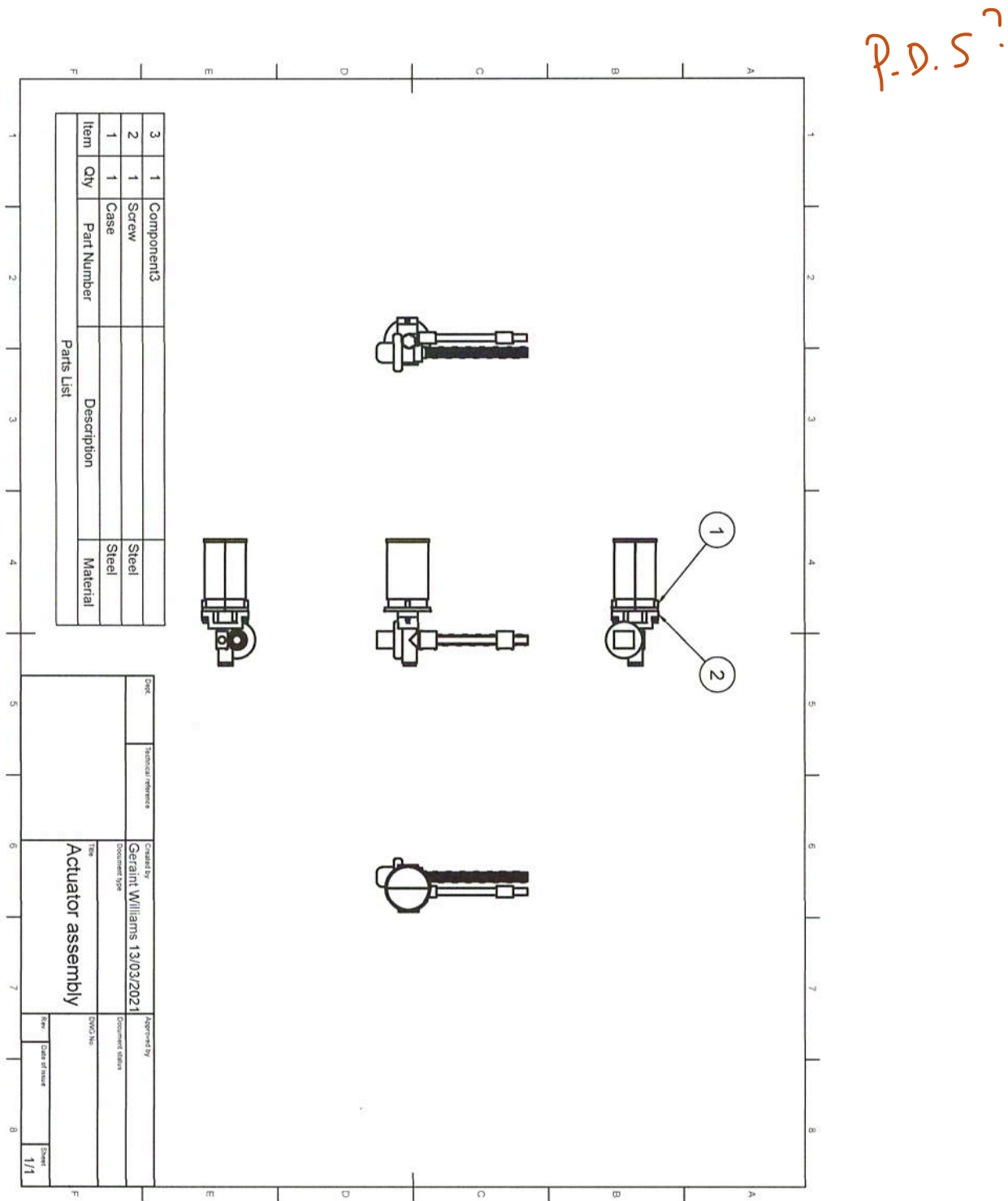


Figure 2: Flap actuator in CAD

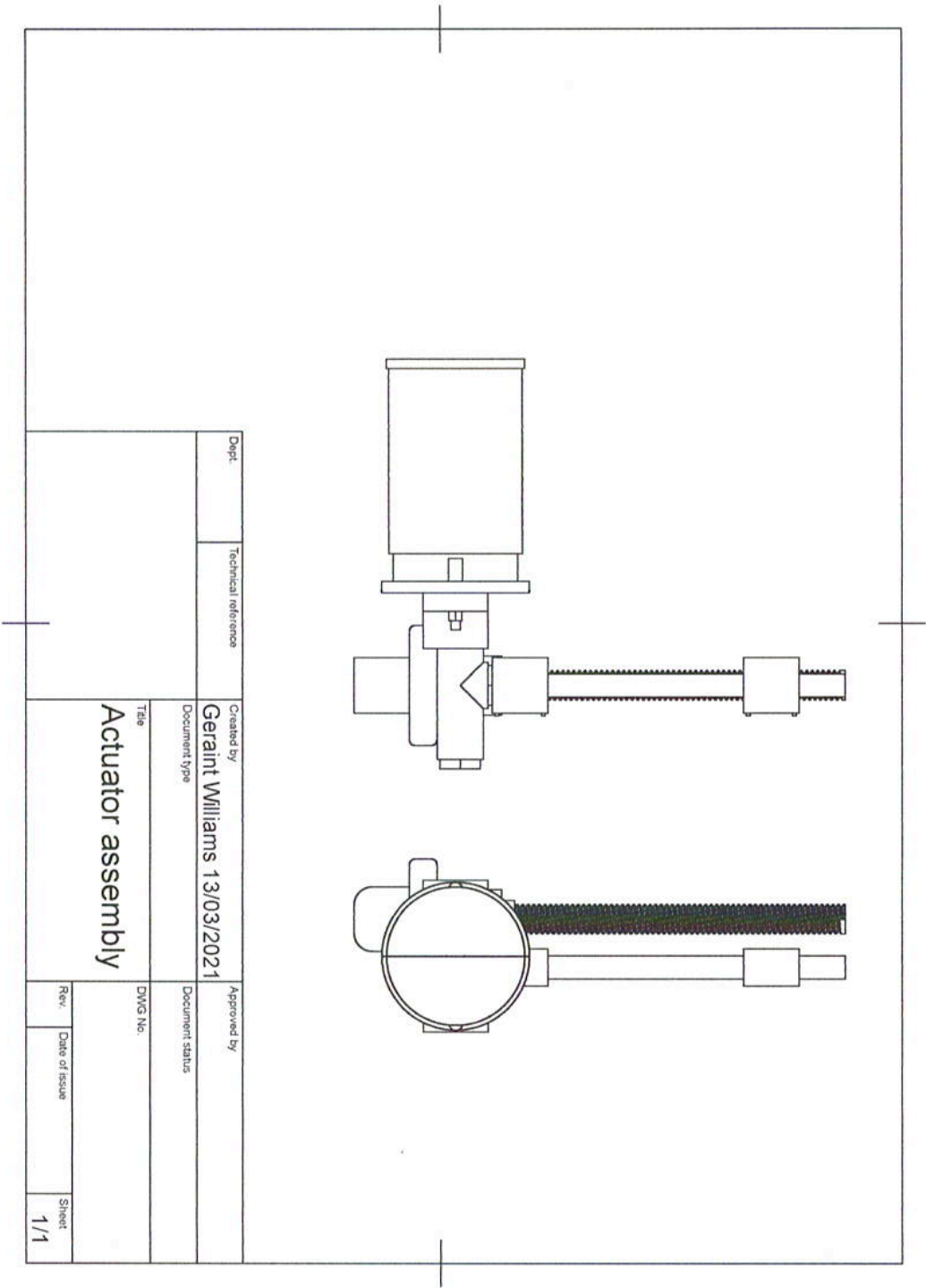


Figure 3: Flap actuator in CAD

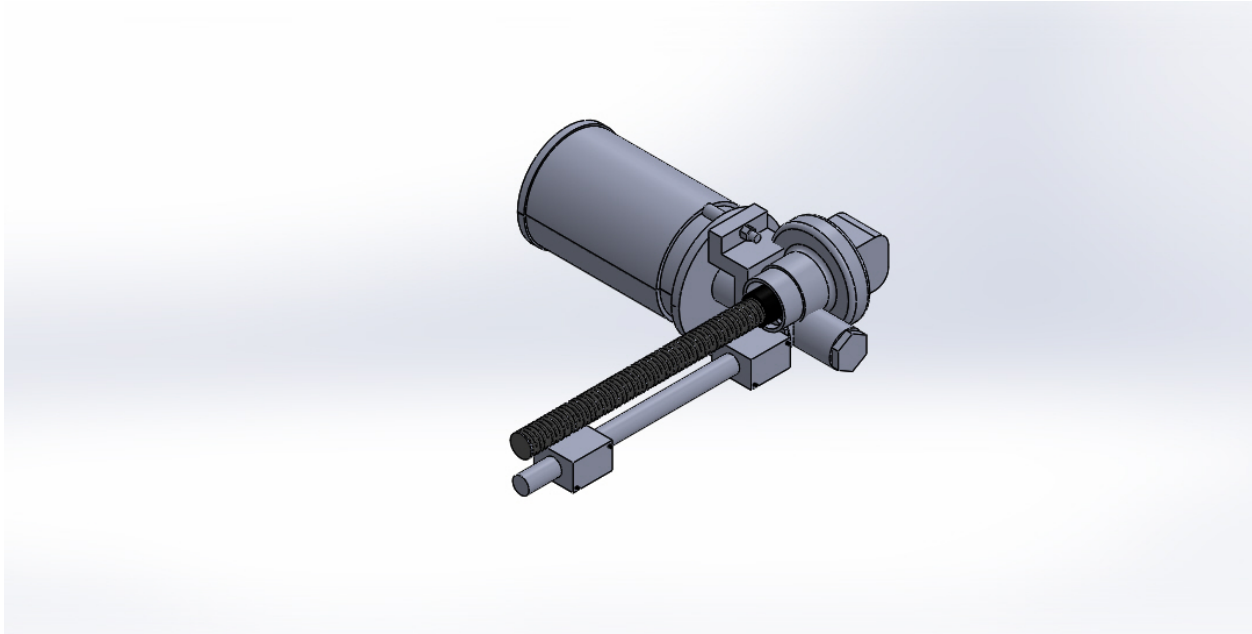


Figure 4: The rendered design of the flap actuator

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data?*

Shaft and Gear analysis, stress and deflection

The shaft and gearbox are crucial aspects of the actuator design. Power and torque transmission within the system is dictated by the configuration of shafts used, material, and arrangement of the gearing system within the system's gearbox. The mode of power transmission for the proposed flap actuator unit from Electromotor is through rotary prime movers; thus, the bearing is essential for the shaft support system. The design utilized steel metal for the shafts systems due to its favoured mechanical properties and spur gears arrangement to necessitate fit power transmission from the motor to the flap's shafts (Taylor, and Panella, 2017, p.56). Whenever power transmission is involved on the shafts, deflection due to bending moments, torsional deflections, and shear deflection due to transverse loading are common occurrences. Thus, the deflection calculation is essential for the design for safety in the proposed flap actuator design.

Bearing selection and life calculations

The selection of bearing used in the mechanical components design is influenced by different performance criteria that include but are limited to required power, the fatigue life of the bearing, temperature limits, rotational precision, and speed calibrations. However, bearing life can be limited by the type of shaft used housing and the system's shaft interface. The bearing life often referred to as the rating life of the bearing, is calculated for 95 percent reliability of the component. The formula for computing bearing rating life is

$$L_{10} = \frac{(C/P)^e \times 10^6}{60 \times N}$$

C = Dynamic Capacity

P = Equivalent Bearing Load

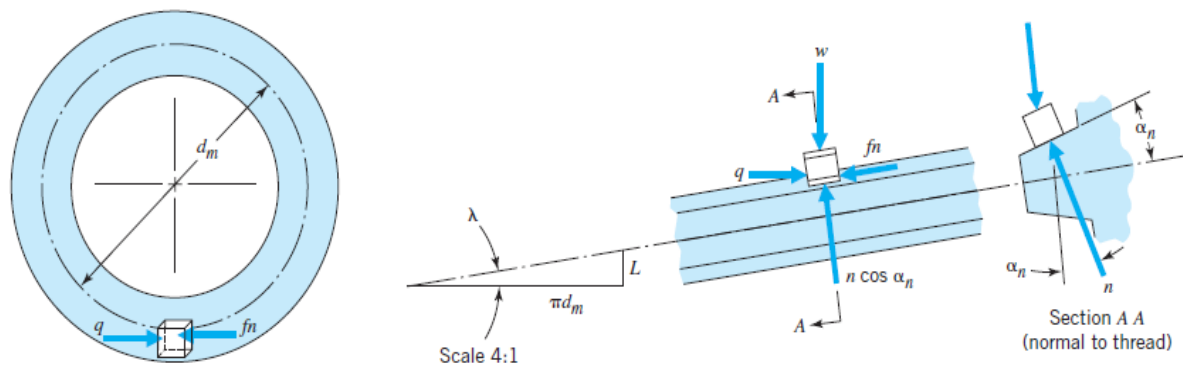
N = Rotating speed in RPM

e = 3.33 for roller bearings and 3.0 for ball bearings

Design calculations

Power, Screw, Linear Actuator Screw Calculations

The main purpose of power screw in the design is to obtain greater mechanical advantage for the system.



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$$\tan \lambda = \frac{L}{\pi d_m}$$

Where

λ = lead angle

L= lead

d_m = mean diameter of thread contact

This finding Lead.

Take $\lambda = 20^\circ$ and $d_m = 70\text{mm}$

$$L = \pi d_m (\tan \lambda) = \pi \times 70 \times \tan 20^\circ$$

$$= 80.04\text{mm}$$

Forces due to:

- Load,
- Axial Load,
- Nominal force,
- Friction forces
- Tangential forces.

Summation of tangential forces.

$$\Sigma F_t = 0: \quad q - n(f \cos \lambda + \cos \alpha_n \sin \lambda) = 0$$

Summation of axial forces.

$$\Sigma F_a = 0: \quad w + n(f \sin \lambda - \cos \alpha_n \cos \lambda) = 0$$

Summation of normal force.

$$n = \frac{w}{\cos \alpha_n \cos \lambda - f \sin \lambda}$$

$$q = w \frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda}$$

Thus

$$T = Q \frac{d_m}{2} = \frac{W d_m}{2} \frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda}$$

$$T = F \times a$$

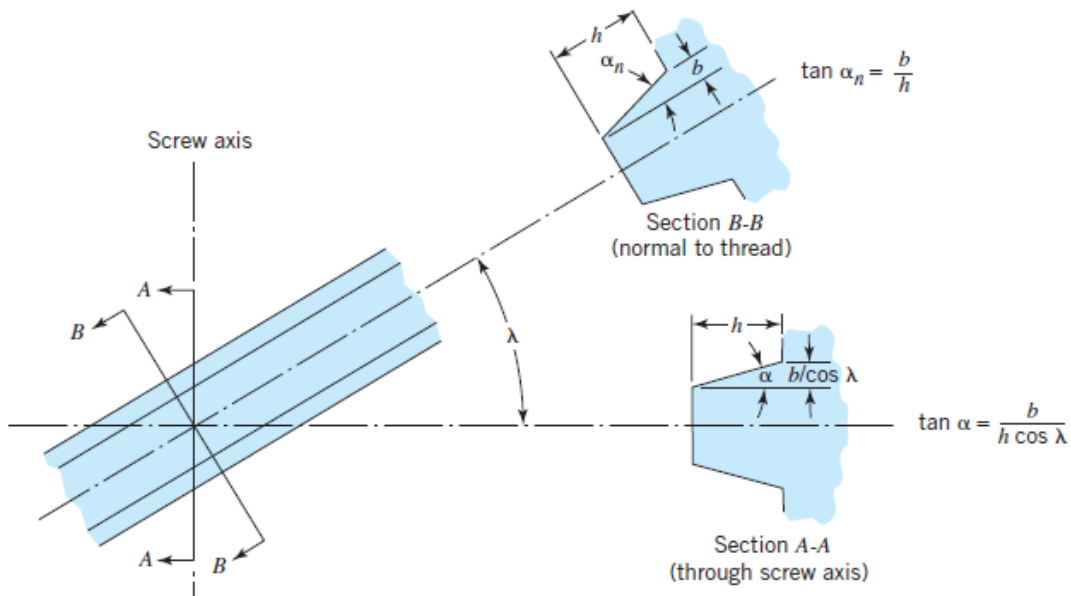
Power screw design to have fine threads per inch.

$$P = 1/5 = 0.2 \text{ in} \simeq 5\text{mm}$$

Due to double thread,

$$L = 2 \times P = 2 \times 5 = 10\text{mm}.$$

$$d_m = d - p/2 = 70 - 2.5 = 67.5\text{mm}$$



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(α and α_n).

$$\tan \alpha_n = \tan \alpha \cos \lambda$$

$$\alpha_n = \tan^{-1}(\tan \alpha \cos \lambda)$$

$$\tan^{-1}(\tan 14.5 \cos 20) = 13.70^\circ$$

$$T = \frac{1000(70) \times 0.4 \cos 20 + \cos 13.70 \sin 20}{2 \cos 13.70 \cos 20 - 0.4 \sin 20}$$

$$= 4698.96 \text{ Nm}$$

Screw Stresses

Torsional stress

$$\tau = \frac{Tc}{J} = \frac{16T}{\pi d r^3}$$

$$= \frac{16 \times 4698.96}{\pi \times 0.5^3} = 119.63 \text{ Gpa.}$$

Force.

$$T = F \times a$$

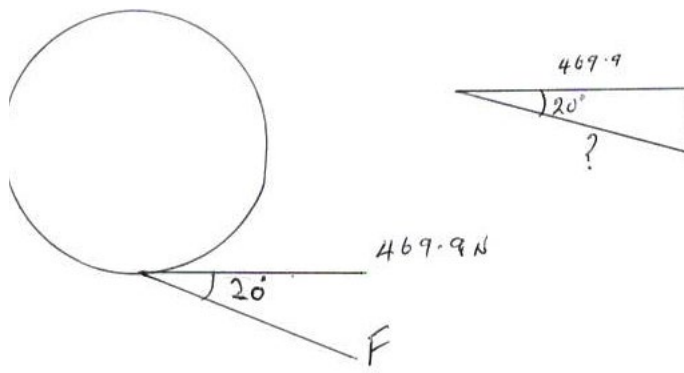
$$= F = T/a$$

$$= \frac{4698.96}{10} = 469.9 \text{ N}$$

EFFICIENCY OF POWER SCREW

$$\eta = \frac{\text{output work}}{\text{input work}} = \frac{w \times L}{2\pi x T} = 73\%$$

Shaft and Gear Calculations



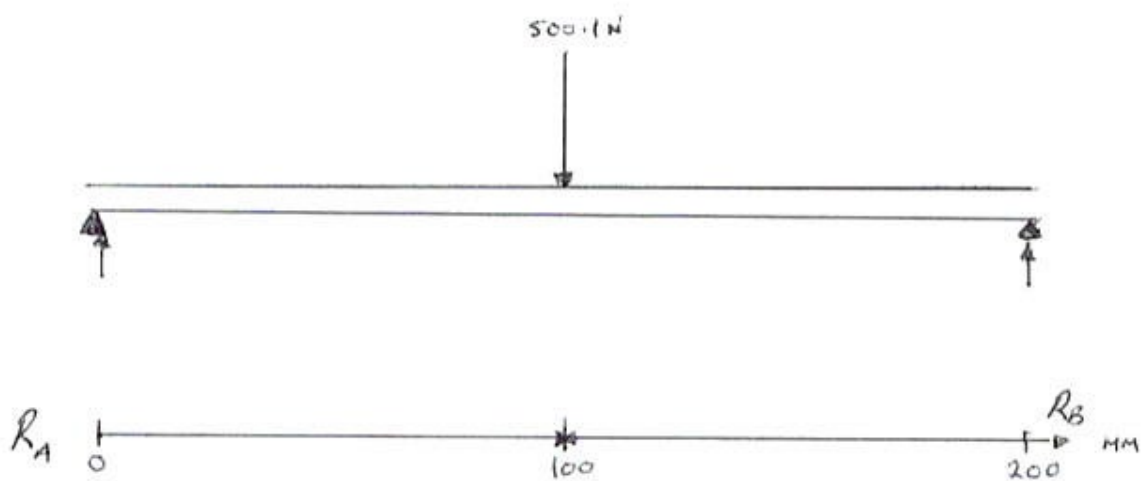
Finding Force on Shaft

$$\cos 20 = \frac{469.9}{F}$$

$$F = \frac{469.9}{\cos 20} = 500.1 \text{ N}$$

Shaft input for input gears

Maximum length of 200mm



$$\sum M = 0 \text{ and } \sum F = 0$$

Taking moments about R_A

$$(500.1 \times 0.1) - (R_B \times 0.2) = 0$$

$$R_B = 250.05 \text{ N}$$

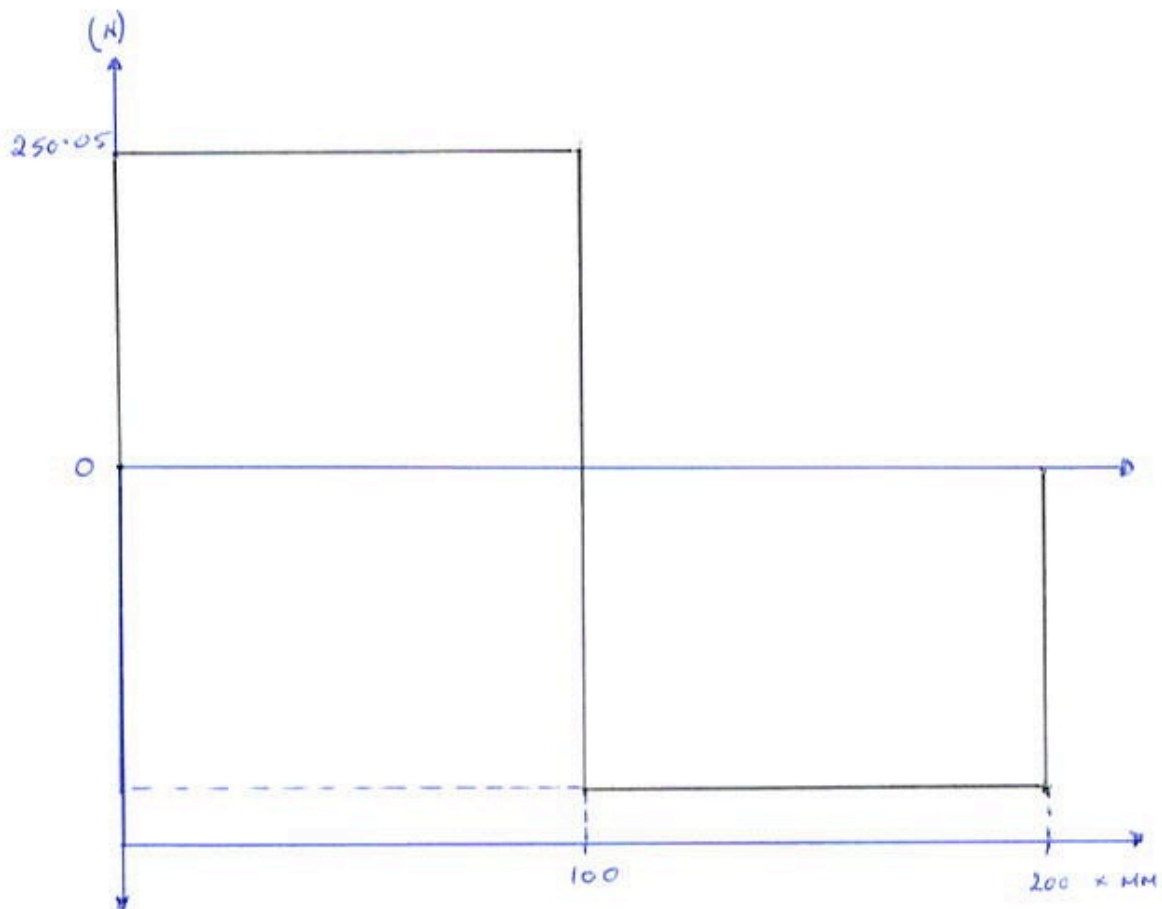
Taking moments about R_B

$$(500.1 \times 0.1) = R_A \times 0.2$$

$$R_A = 250.05 \text{ N}$$

Shear Force Diagram

*This is not
relevant to
Assignment ①?*



Bending Moment

Maximum Bending Moment at 100mm.

$$M = F \times D$$

$$= 250.05 \times 0.1$$

$$= 25.01 \text{ Nm.}$$

Shaft Calculations

From ACME equations

$$d^3 = \frac{32 \times F \times S}{\pi(1-k)\delta y} \sqrt{(K_b m)^2 + (K_t T)^2}$$

Reference from ACME table

$$K = 0.2 \quad K_b = 1.5 \quad K_t = 1.0$$

$$\text{And } \sigma_y = 265 \text{ MPa}$$

Therefore, diameter of Shaft =

$$d^3 = \frac{32 \times 4}{(\pi(1-0.2)(265 \times 10^6))} \sqrt{(1.5 \times 25.01)^2 + (1 \times 23.1)^2}$$

$$d^3 = 6.8469 \times 10^6$$



$$D = 0.019 \text{ m} \approx 19\text{mm}$$

Deflection and slope of Shaft

$$\theta = \frac{FL^2}{16EI} \quad I = \frac{\pi D^4}{64} \quad \text{and} \quad Y = \frac{Fr}{48EL}$$

$$\theta = \frac{500.1 \times 0.2^2}{(16 \times 205 \times 10^9) \times I}$$

$$I = \frac{\pi \times 0.019^4}{64} \quad 6.397 \times 10^{-9}$$

$$\theta = \frac{500.1 \times 0.2^2}{(16 \times 205 \times 10^9) \times (6.397 \times 10^{-9})}$$

$$= 9.534 \times 10^4 \text{ rads}$$

Deflection

$$Y = \frac{FL^3}{48ET} = \frac{500.1 \times 0.2^3}{48(205 \times 10^9)(6.397 \times 10^{-9})}$$

$$= 3.051 \times 10^{-3} \text{ m} = 3.05\text{mm}.$$

Gear Speeds and Gear Calculations.

Gearbox Sizing

Max dimensions $200 \times 300 \times 200$

Input Speed 1500 rpm

Output Speed 500 rpm

$$\text{Train Value } e = \frac{\text{rpm out}}{\text{rpm in}} = \frac{500}{1500} = 0.3$$

In other relation

$$e = \frac{\text{inner dia}}{\text{outer dia}}$$

For the gear to mesh, assuming that total dia of 150mm

$$\text{If } d_1 + d_2 = 150\text{mm}$$

$$d_1 = 55\text{mm}$$

$$d_2 = 95\text{mm}$$

Number of Teeth

From the Module = 3mm

$$M = \frac{d}{n}$$

$$\text{Input Teeth} = \frac{dim}{mod} = \frac{55}{3} = 19 \text{ Teeth}$$

$$N_{in} = 19$$

$$N_{out} = 95/3 = 32 \text{ Teeth}$$

Torque on Gears

$$T = \frac{\text{Angular Velocity}}{\text{Power}} = \frac{\omega}{\rho} \frac{\rho}{\omega}$$

Small Gear

$$\text{Input Power} = 3000 \omega$$

$$\omega = 1500 \times \frac{2\pi}{60} = 157 \text{ rads}$$

$$T = \frac{300}{157} = 19.12 \text{ Nm}$$

Output Torque

$$\frac{T_{in}}{d_{in}} = \frac{T_{out}}{d_{out}}$$

$$\frac{55}{95} = \frac{19.12}{d}$$

$$\frac{55}{95} = T_{out}$$

$$T_{out} = \frac{T_{in} \times d_{out}}{d_{in}} = \frac{19.12 \times 95}{55} = 33.03 \text{ Nm}$$

Tangential Forces

$$T = \frac{d}{2} W_t \quad \text{Therefore } W_t = \frac{2T_{in}}{d_{in}}$$

$$W_t = \frac{2 \times 33.03}{0.095}$$

$$= 695.37 \text{ N}$$

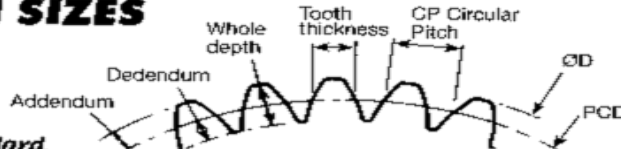
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Appendixes

GEAR TOOTH SIZES



Bold numbers are standard.
Dimensions in mm and inches.

MOD	DP	CIRCULAR PITCH	TOOTH THICKNESS	ADDENDUM (= MODULE)	DEDENDUM	WHOLE DEPTH
0.2 MOD	127.00	0.628	0.314	0.20	0.28	0.48
0.25 MOD	101.60	0.785	0.392	0.25	0.35	0.60
0.254	100 DP	0.0314"	0.0157"	0.010"	0.014"	0.0240"
0.265	96 DP	0.0327"	0.0163"	0.0104"	0.0145"	0.0250"
0.3 MOD	84.67	0.942	0.471	0.30	0.42	0.72
0.318	80 DP	0.0392"	0.0196"	0.0125"	0.0175"	0.0300"
0.353	72 DP	0.0436"	0.0218"	0.0138"	0.0194"	0.0333"
0.397	64 DP	0.0490"	0.0245"	0.0155"	0.02187"	0.0375"
0.4 MOD	63.50	1.256	0.628	0.40	0.56	0.96
0.5 MOD	50.80	1.570	0.785	0.50	0.70	1.20
0.529	48 DP	0.0654"	0.0327"	0.0208"	0.02916"	0.0500"
0.6 MOD	42.33	1.884	0.942	0.60	0.84	1.44
0.635	40 DP	0.0785"	0.0392"	0.025"	0.0350"	0.0600"
0.7 MOD	36.29	2.199	1.099	0.70	0.98	1.68
0.706	36 DP	0.0872"	0.0436"	0.0277"	0.0388"	0.0666"
0.75 MOD	33.87	2.356	1.178	0.75	1.05	1.80
0.794	32 DP	0.0981"	0.04906"	0.0312"	0.0437"	0.0750"
0.8 MOD	31.75	2.513	1.256	0.80	1.12	1.92
0.847	30 DP	0.1047"	0.05235"	0.0333"	0.0466"	0.08"
0.9 MOD	28.22	2.827	1.413	0.90	1.26	2.16
0.907	28 DP	0.1121"	0.05605"	0.0357"	0.0500"	0.0857"
1.00 MOD	25.40	3.141	1.570	1.00	1.400	2.40
1.05	24 DP	0.1308"	0.0654"	0.0416"	0.0583"	0.1000"
1.15	22 DP	0.1427"	0.07135"	0.0454"	0.0636"	0.1090"
1.25 MOD	20.32	3.926	1.963	1.25	1.562	2.81
1.27	20 DP	0.1570"	0.0785"	0.050"	0.0625"	0.1125"
1.411	18 DP	0.1745"	0.08725"	0.0555"	0.0694"	0.1250"
1.50 MOD	16.33	4.712	2.356	1.50	1.875	3.37
1.588	16 DP	0.1963"	0.09815"	0.0625"	0.0781"	0.1406"
1.693	15 DP	0.2094"	0.1047"	0.0666"	0.0833"	0.1500"
1.75 MOD	14.51	5.497	2.7485	1.75	2.187	3.93
1.814	14 DP	0.2243"	0.11215"	0.0714"	0.0892"	0.1607"
2.0 MOD	12.70	6.283	3.141	2.00	2.50	4.50
2.117	12 DP	0.2617"	0.13085"	0.0833"	0.1041"	0.1875"
2.25 MOD	11.29	7.068	3.534	2.25	2.812	5.06
2.5 MOD	10.16	7.853	3.926	2.50	3.125	5.62
2.54	10 DP	0.3141"	0.15705"	0.100"	0.1250"	0.2250"
2.75 MOD	9.24	8.639	4.319	2.75	3.437	6.18
3 MOD	8.47	9.424	4.712	3.00	3.75	6.75
3.175	8 DP	0.3926"	0.1963"	0.125"	0.15625"	0.28125"
4 MOD	6.35	12.566	6.283	4.00	5.00	9.00
4.233	6 DP	0.5235"	0.26175"	0.1666"	0.20833"	0.3750"
5 MOD	5.08	15.707	7.853	5.00	6.25	11.25
5.08	5 DP	0.6283"	0.31415"	0.200"	0.2500"	0.4500"
6 MOD	4.23	18.849	9.4245	6.00	7.50	13.50
6.35	4 DP	0.7853"	0.39265"	0.250"	0.3125"	0.5625"
8 MOD	3.175 DP	25.132	12.566	8.00	10.00	18.00

762 Unit 14, Foxwood Industrial Park, Foxwood Road, Chesterfield, Derbyshire S41 9RN
Telephone +44(0)1246 268080 Fax +44(0)1246 260003

Figure 5: Table of gear sizing